Waste heat recovery system for a generation IV nuclear reactor spent fuel storage tank using natural circulation and two-phase closed-loop thermosyphon-type heat pipes

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Abstract
From a reliability and safety point of view the use of entirely passive natural circulation loops for transferring heat is most desirable, and particularly in nuclear reactors applications where safety is of paramount importance. This paper focuses on the design, construction, simulation and experimental evaluation of a one-twentieth scale cooling and heat recovery system for a generation IV used/spent fuel storage tank. Full scale used/spent fuel tank would typically be 18 m high and three meters diameter made of carbon sealed steel, with a wall thickness of approximately 16 mm, capable of sustaining one MPa internal pressure and allowed to maintain a temperature of about 250 °C for an extended period of time of at least 40 years before being removed from the reactor building and transported to a permanent storage site. Each spent fuel releases an amount of heat caused by the radioactive decay of unstable fission products inside the used/spent fuel. Unless revalorised for economically beneficial purposes, this heat is wasted.

Key Words: Two-phase closed loop thermosyphon type heat pipes, entirely passive heat transfer systems, Generation IV reactor fuel storage tank cooling

1. Introduction
Waste heat recovery systems (WHRSs) conserve energy by storing and reusing available waste heat. They transfer energy from sources of waste heat to energy conversion systems, by using various types of heat recovery equipment. They reduce energy consumption which results in significant cost savings (Reiter, 1983). WHRSs form an important part of the methods developed to increase the efficiency of various energy saving systems.

WHRS equipment processes waste heat at any temperature from chilled cooling water to the high-temperature waste heat of a nuclear reactor (Fujima et al., 2000). Usually high-temperature waste heat leads to more efficient energy recovery and cost effectiveness. Typical examples of WHR techniques are preheating of fuel-air mixtures, space heating and pre-heated boiler feed water or process water. For high-temperature energy recovery, a cascade of waste heat recovery equipment may be used to ensure that the maximum amount of heat is recovered. An important application of waste heat equipment is in the use of high-temperature waste heat for air preheating or low-temperature waste heat for process feed water heating or steam generation (Reiter, 1983).

One of the unique features of the generation IV (G-IV) reactor is the TRISO fuel used for the fission reaction. Most of the generation IV reactors use small fuel particles to accomplish the nuclear reaction. The fuel is re-cycled through the reactor several times in its lifetime until it is partially burned and/or optimally used-up. Partially burnt fuel may be removed for recycling while spent fuel will be removed from the reactor. The highly radioactive fuel removed will be stored in storage tanks. As seen in figure 1, most storage tanks are large
cylindrical pressure vessels made of carbon sealed steel capable of sustaining one MPa internal pressure and temperature of 250 °C. Inside the tank are a number of cooling tubes running from the bottom to the top. The fuel spheres are stored in an inert gas environment at atmospheric pressure. Each spent fuel sphere releases an amount of heat caused by the radioactive decay of unstable fission products inside the sphere. The amount of heat greatly depends upon the time that has elapsed from the moment the nuclear fission reaction has stopped. No heat is actively removed from inside the tank (Fuls et al., 2005).

![Diagram of storage tank layout](Figure 1: Storage tank layout (Fuls and Mathews, 2006))

2. Objectives and design considerations

2.1 Objectives

This paper focuses on heat transfer and heat conversion by means of passive cooling processes using heat pipes and heat exchangers to construct a WHRS for a G-IV reactor fuel tank. The main aim is to develop conceptual, design and modelling theories to construct a WHRS that will convert the recovered thermal energy into a useful form of energy, such as electricity, by means of an energy conversion system. The WHRS design and modelling theory developed in this paper should comply with the nuclear safety requirements.

Safety is the main concern in nuclear reactor engineering particularly in the WHRS of G-IV reactor. Safety emphasises the concept of defence-in-depth. Defence-in-depth requires that more than one totally independent cooling system is available and capable of cooling the fuel tank should one or the other not function. To achieve this, two lines regulated automatically by a thermostatically controlled valve are used in this design. This valve will reduce the risk that a primary line failure could interfere with the cooling process of the fuel tank. Using a back-up natural circulation and convection in an air-cooled condenser, the design allows the cooling process to continue automatically should a failure in the primary cooling process occur.

2.2 Design considerations

G-IV spent fuel storage tanks are thermally hot as well as highly radioactive, hence it is required that the temperature of the fuel within the tank should be maintained at temperatures \( \leq 400 \) °C by cooling (Fuls and Mathews, 2006). However the carbon steel tank must be kept below about 250 °C to ensure a proper fuel tank cooling (Slabber 2008). A natural circulation WHRS for a G-IV reactor spent fuel tanks will conserve energy by storing and reusing the available waste heat. They will naturally transfer thermal energy from the storage tanks to energy conversion systems, by using a cooling process in a particular form of heat pipes called a closed-loop thermosyphon. WHRSs are totally dependent on the temperature of the waste heat and the nature of the working fluid used in the thermosyphon loop.
Figure 2 shows the schematic diagram of the two-phase flow WHRS that will be considered. It shows how waste heat will be recovered, transported and transferred from the fuel storage tanks to where it will be reutilised in a process that will produce electricity though an appropriate energy conversion unit. Waste heat from a fuel storage tank will be recovered by submerging the storage tank into a cylindrical container filled with water. The spent fuel will transfer its decay heat through the tank wall to the water surrounding the tank in the cooling jacket. Steam is produced in the space above the cooling jacket water level. The fuel temperature in the tank is then reduced to a temperature slightly exceeding the cooling jacket operating temperature dictated by the saturated steam pressure. Thus, the cooling jacket will simply play the role of a vapour generator.

Waste heat contained in the steam produced in the cooling jacket is transported to the heat exchanger by means of natural circulation in a particular type of heat pipe called the two-phase closed-loop thermosyphon. In a natural circulation loop, the fluid flow is driven by the buoyancy forces arising from the density gradient in the working fluid between the heating and the condenser sections of the loop. There are no moving parts needed to induce the fluid circulation in the loop (Mori et al., 1993). The only driving force source is the heat input and output to the system. When heat is gradually added to a fluid in the cooling jacket, the fluid temperature will increase by convection until it reaches a temperature slightly above the fluid saturation temperature, from which nucleation and boiling can commence. During nucleation and boiling, the fluid flow will be characterised by two phases, namely vapour and liquid (Dobson, 1993). The number of phases of the working fluid defines the operating mode of the thermosyphon. Because of natural means and no use of moving parts, this system is relatively noiseless and also less likely to fail. It eliminates the cost and maintenance problems of pumps, as in forced convection systems. However, one of the major problems in natural circulation systems is the occurrence of various types of instabilities. Instabilities are known to occur depending on the system geometry and operating conditions.

Figure 2 shows the two main lines of the thermosyphon loop, which do not work at the same time. The primary line comprises a cooling jacket and a heat exchanger, while the secondary line comprises a cooling jacket and a natural convection air-cooled condenser. This is also an emergency fail-safe line, to ensure that the fuel tank is cooled when the power conversion unit has to be switched off for maintenance, or if it fails. The thermostatically controlled valve provides a reliable, automatic control of the operating pressure of the system and regulates the work of a specific line according to the working conditions of the system. The system working conditions are dependent on the primary line working pressure. An increase in pressure or temperature in the primary line due, for example, to a sudden failure
of the heat exchanger will be sensed by the thermostatically controlled valve, which will then open the secondary line and the fuel tank cooling process continues without interruption. Two-phase flow heat exchangers are used to save energy by direct condensation of a vapour into a liquid. During direct condensation of a vapour into a liquid energy is stored in the cooling jacket. When energy is needed again, the liquid is depressurised and flashing occurs, which results in vapour production. Not only does condensation reduce the volume of working fluid that has been supplied and treated, but the heat contained in the condensate is also recovered.

### 2.3 Energy conversion device

The energy efficiency of a G-IV reactor may be increased by using an energy conversion device in a waste heat recovery and utilisation system WHR&US to capture and reutilise much of the reactor waste energy. The efficiency of a WHR&US is directly dependent on the temperature range of the heat source. Figure 3 shows the simulation results of the thermal efficiency for the three identified energy conversion systems in terms of different temperatures of waste heat sources.

For some specific temperature ranges, the following energy conversion systems have been identified as appropriate for a G-IV reactor WHR&US. For \( T \leq 100 \degree C \): a dual function absorption cycle (DFAC). For \( 100 \degree C \leq T \leq 350 \degree C \): an organic Rankine cycle (ORC). For \( 350 \degree C \leq T \leq 600 \degree C \): Stirling cycle (SC).

![Figure 3: Thermal efficiency as a function of heat source temperature](image)

### 3. Mathematical modelling

Due to numerical instabilities in two-phase flow systems, the mathematical model is simplified assuming a one-dimensional, incompressible (for both the liquid and vapour phases) and quasi-static flow. This implies that the dynamic behaviour of the flow in the loop may be analysed as a steady-state flow at each time step (Aritomi et al., 1993; Dobson, 1993). However, since the density of the vapour changes with temperature, this assumption introduces some errors. These errors are offset by the simplicity this assumption offers for the solution algorithm. Welander (1967) suggests that the following assumptions should also be made:

- The Boussinesq approximation.
- The tangential force on the working fluid is proportional to the square of the flow rate.
- The temperature of the working fluid in the thermosyphon pipe is uniform over its cross section.

The Boussinesq approximation is used in the theoretical model of the thermosyphon and it assumes that, the density and the viscosity variations are small and affect only the force terms of the momentum balance equation (Agrawal, 2006). The reason why the density cannot be assumed to be a constant in the buoyancy term of the momentum equation is because the
buoyancy is the dominant force that drives the flow in the loop. The density varies with temperature as \( \rho = \rho_{ref}(1 + \beta(T - T_{ref})) \), where \( \beta \) is the thermal volumetric expansion rate of the fluid at density \( \rho_{ref} \) and temperature \( T_{ref} \) in °C.

### 3.1 Conservation equations

The conservation equations, written according to the laws of mechanics are applied to each specific control volume as shown in figure 4, with all the fluid properties located at the centre of the control volume (Dobson, 2007).

![Figure 4: Conservation equations control volumes, (a) Momentum, (b) Energy and mass](image)

Conservation of mass (see figure 4(b))

\[
\frac{\Delta m}{\Delta t} = \dot{m}_{in} - \dot{m}_{out}
\]

Conservation on momentum (see figure 4(a))

\[
\frac{\Delta}{\Delta t} (\rho \vec{v}) = (\rho \vec{v})_{in} - (\rho \vec{v})_{out} + \sum F_{sys}
\]

Conservation of energy (see figure 4(b))

\[
\frac{\Delta}{\Delta t} (\rho c_{p} T) = (\rho c_{p} T)_{in} - (\rho c_{p} T)_{out} + Q
\]

### 3.2 Heat transfer equations

Heat is removed from the fuel tank by the thermosyphon through the cooling jacket. The working fluid transport within the thermosyphon is due to the heat transfer process taking place in heat exchangers. An electrical resistance analogy is used to illustrate the different heat transfer equations as (Mills, 1999).

![Figure 5: Heat removed from a thermosyphon section](image)

The heat removed from a thermosyphon section is

\[
Q_{ij} = \frac{T_{i} - T_{j}}{R_{ij}}
\]

where, the air-cooled condenser thermal resistance \( R_{ij} = R_{co} + R_{co,wp} + R_{th} \), with the convective thermal resistance between the working fluid and the pipe inner and outer area.
with \( j = i, o \) are \( R_{coj} = \left( \frac{2 \pi L \cdot d_{j} \cdot \ln \left( \frac{d_{j} + 2 \tau_{j}}{d_{j}} \right)}{2 \pi L \cdot d_{j} \cdot k_{j}} \right)^{1} \). The pipe wall resistance is

\[
R_{co,w} = \ln \left( \frac{d_{p} + 2 \tau_{p}}{d_{p}} \right) \left( \frac{2 \pi L \cdot d_{p} \cdot k_{j}}{2 \pi L \cdot d_{p} \cdot k_{j}} \right)^{1}.
\]

The air-cooled condenser air side thermal resistance is

\[
R_{th} = \left[ \frac{1}{R_{co,co}} + \frac{1}{R_{co,co}} + \frac{1}{R_{co,co}} \right]^{1} \tag{3.2.2}
\]

Figure 6: Heat removed through air-cooled condenser

The radiation thermal resistance as proposed by Mills (1999) is

\[
R_{rad} = \frac{1}{\sum_{i=1}^{N} \left( \frac{d_{i}}{A_{i}} \right)^{2} + \frac{1}{A_{i} \cdot \ln \left( \frac{d_{i}}{\ln \left( \frac{d_{i}}{d_{i}} \right)} \right)}} \tag{3.2.3}
\]

The heat exchanger thermal resistance

\[
R_{hj} = \sum_{i=1}^{N} \left( \frac{d_{i}}{A_{i}} \right)^{2} \cdot R_{h,j} \tag{3.2.4}
\]

where, \( R_{h,j} = \frac{1}{\pi d_{j} \cdot k_{j}} \).

Considering the quasi-static and incompressible flow in the integration of equations (3.1.1) and (3.1.2) around the loop in the interval \([t, t + \Delta t]\), and for each control volume \( k \), the actual mass flow rate is:

\[
m_{1k}^{\Delta t} = \frac{\Delta t}{\sum_{i=1}^{N} \left( \frac{d_{i}}{A_{i} \cdot \ln \left( \frac{d_{i}}{d_{i}} \right)} \right)} - \sum_{i=1}^{N} \frac{\Delta P_{f_i}}{\rho_{i} \cdot A_{i} \cdot k_{i} \cdot h_{i}} \tag{3.2.5}
\]

where,

\[
M_{1} = \sum_{i=1}^{N} \frac{\Delta t}{A_{i} \cdot \ln \left( \frac{d_{i}}{d_{i}} \right)} \left( \frac{1}{\rho_{i} \cdot A_{i}} - \frac{1}{\rho_{i} \cdot A_{i}} \right) + \sum_{i=1}^{N} \rho_{i} \cdot A_{i} \cdot h_{i} \cdot \sin \theta_{i}
\]

\[
\sum_{i=1}^{N} \Delta P_{f_i} = \sum_{i=1}^{N} \frac{1}{2} \left( \frac{C_{f_j} + h_{j}}{\rho_{j}} \right) \frac{\left( m_{1} \right)^{2}}{A_{1}^{2}}
\]

The following closure equations where used for a single phase control volume

\[
c_{f_j} = 1 \text{ \ if \ } Re_{j} \leq 1
\]

\[
c_{f_j} = \frac{1}{Re_{j}} \text{ \ if \ } 1 \leq Re_{j} \leq 1181
\]

\[
c_{f_j} = 0.0791 Re_{j}^{-0.25} \text{ \ if \ } Re_{j} > 1181
\]

Where,

\[
Re_{j} = \frac{4 \cdot m_{1} \cdot d_{j} \cdot \sin \theta_{j}}{\pi \cdot d_{j} \cdot h_{j}}
\]
The coefficient of friction of a two-phase control volume is assumed as the single-phase coefficient of friction multiplied by the so-called two-phase flow frictional multiplier $c_{ij} = c_i c_j^{1.75}$. The two-phase frictional multiplier may be given, assuming turbulent flow for both the single- and two-phase, in terms of the so-called Martinelli as:

$$
\phi_{ij} = \left( \frac{1-x}{0.15} \right)^{0.85} \left( \frac{1}{x} \right)^{0.85}
$$

where $\phi_{ij}$ is the two-phase frictional multiplier.

The two-phase flow density may be given by the Lockhart-Martinelli correlation (Kakac and Liu, 2002) as:

$$
\rho = \alpha \rho_v \left( 1 - \alpha \right) \rho_l
$$

and the void fraction by $\alpha = \left( 1 + 0.28 \right) \left( \frac{0.71}{y} \right)^2$, where the two phase properties may be approximate as: $\rho_l = \alpha \rho_v + (1-\alpha) \rho_l$, $k = \alpha k_v + (1-\alpha) k_l$, and $c = \alpha c_v + (1-\alpha) c_l$.

The integration of equation (3.1.3) allows the computation of the working fluid temperature in each control volume and the thermodynamic quality of the working fluid as follow.

$$
u = \left[ \frac{\nu_i}{\nu} + \frac{(1-x)}{\nu} \right] - \phi
$$

where, $\phi$ is positive when heat is added to and negative when removed from the control volume.

If $\nu_i = \nu$ then $T_{n+1} = T_n$ and $x_{n+1} = 0$

If $\nu_i > \nu$ then $T_{n+1} = T_{n+1} - T_{n-1}$

and $x = \left( \frac{u_{n+1} - u_{n-1}}{u_{n+1} - u_{n-1}} \right)

x$

where, $u_{n+1} = c_v T_{n+1}$, therefore the general form of equation (3.2.6) becomes

$$
T_{n+1} = T_n + \frac{\Delta t}{m c_v} [\Delta t (h_{n+1} - h_{n+1})] \pm \phi
$$

Applying equation (3.2.7) to the cooling jacket, the initial temperature assumed to be the temperature of the spent fuel in the fuel tank is known. The temperature of the working fluid in the thermosyphon heating section at the next time step may be computed as

$$
T_{n+1} = T_n + \frac{\Delta t}{m c_v} \left[ \Delta t (h_{n+1} - h_{n+1}) \right] - \Delta t h_{t} - h_{t} - h_{t}
$$

Applying equation (3.2.7) to an air-cooled condenser and/or the heat exchanger control volume as shown in figure 12, Where the ambient and/or coolant temperature is know, it may be seen that the temperature of the working fluid at the next time step may be found as:

$$
T_{n+1} = T_n + \frac{\Delta t}{m c_v} \left[ \Delta t (h_{n+1} - h_{n+1}) \right] - \Delta t h_{t} - h_{t} - h_{t}
$$

4. Experimental and theoretical results

Figure 7 shows a schematic diagram of a thermosyphon loop designed and constructed to demonstrate the recovery of waste heat from a G-IV reactor spent fuel storage tank. The thermosyphon loop consists of a primary and secondary line, manually regulated by four shut-off valves, $V_2$, $V_3$, $V_8$, and $V_9$. The primary line comprises the cooling jacket and the heat exchanger, while the secondary line comprises the cooling jacket and the air-cooled condenser. The cooling jacket is concentric to a fuel tank containing electrical heating elements that are used to simulate the heat transferred to the cooling jacket through the fuel tank walls. The heated-up fuel tank walls will raise the temperature of the water in the cooling jacket. If the temperature of water is raised beyond its saturation point, then vapour is produced. The flow in the loop will then be characterised by a two-phase mixture of vapour
and liquid. Therefore, three modes of operation, namely single phase, single to two-phase, and heat pipe mode, may be considered.

4.1 Summary of sensitivity analysis

In order to validate the WHR&U system theoretical model, a sensitivity analysis was undertaken to investigate the more important parameters affecting the theoretical results. The experimental results were set as a reference base line for the theoretical model. It was necessary to analyse in which way changes in the flow and geometrical variables of the cooling sections of the WHR&U system loop affect theoretical results. The flow variables considered in this analysis are the air and working fluid mass flow rate ($\dot{m}_a$, $\dot{m}_w$), dynamic viscosity ($\mu_a$, $\mu_w$), density ($\rho_a$, $\rho_w$), thermal conductivity ($k_a$, $k_w$) and specific heat constant ($c_p$, $c_v$). The geometrical variables considered are the heat transfer area ($A_t$), the diameter of the heat exchanger tubes ($d_t$), the number of tubes ($N_t$), the tube transversal pitch ($s_t$), the tube longitudinal pitch ($s_L$), and the tube layout. It was found that, the flow and geometrical variables influencing the theoretical results are captured in the thermal resistances of the air-cooled condenser and heat exchanger, respectively. It was then, convenient to investigate how changes in the air-cooled condenser and heat exchanger thermal resistance affect the theoretical results. Yang et al. (2006) suggest that, in cases like this, the theoretically determined total thermal resistance may be adjusted accordingly as

$$R_{\text{tot,actual}} = R_{\text{tot,ideal}} R_a^{n_a}$$ (4.1.1)

In equation (4.1.1), $R_{\text{tot,actual}}$ is the actually determined total thermal resistance, $R_{\text{tot,ideal}}$ is the theoretically determined total thermal resistance, $R_a$ the experimentally determined Raleigh number, and $n_a$ is an experimentally determined constant. In this case the experimentally determined total thermal resistance $R_{\text{tot,actual}}$ is assumed to be the actual total resistance $R_{\text{tot,actual}}$. 

Figure 7: Schematic diagram of the experimental set-up
4.2 Discussion of results

Figure 8 and 9 show a typical single to two-phase mode experimental, theoretical, set of results for the air-cooled condenser line. Figure 9 shows the air-cooled condenser working fluid and coolant inlet and outlet temperatures, \( T_{\text{acc}} \), \( T_{\text{coo}} \), \( T_{\text{acc-coo}} \) and \( T_{\text{coo-acc}} \) as a function of time. Figure 9 shows the inlet and outlet cooling jacket fluid temperatures, \( T_{\text{clj}} \) and \( T_{\text{clj-o}} \) and the fuel tank wall temperatures, \( T_{\text{fsw}} \) as a function of time. Figure 8 shows that, the \( T_{\text{acc}} \) increases rapidly to 80 °C within 2 hours. Thereafter the temperature increases slowly to 92 °C after 4 hours when steady state is reached. The \( T_{\text{acc}} \) reached steady state 2 hours later its corresponding predicted value because the power input to the system was effectively removed from the system by the air-cooled condenser. The later was initially designed to remove from the system a power input in the range 3-6 \( kW \). The power input to the system was set to its maximum constant value in the theoretical model, but it increased slowly during experiments. Therefore, the working fluid temperature takes a long time (2 hours) to reach its saturation temperature in the experimental results.

![Figure 8: Single to two-phase mode air-cooled condenser temperatures.](image_url)

![Figure 9: Single to two-phase mode air-cooled condenser line fuel tank wall and cooling jacket temperatures.](image_url)

Figure 9 shows that the cooling process of the fuel tank is sustained by the bulk fluid temperature at the inlet \( T_{\text{clj}} \) and outlet \( T_{\text{clj-o}} \) points of the cooling jacket, it keeps \( T_{\text{fsw}} \) at about 215 °C. The \( T_{\text{clj}} \) and \( T_{\text{clj-o}} \) were accurately predicted. The cooling process undertaken by the presence of water in the cooling jacket keeps the fuel tank temperature around 215 °C. This is due to heat removed from the system by the air-cooled condenser. These results show that, by keeping the temperature of the fuel tank wall around 215 °C, its cooling process is achieved.

![Figure 10: Single to two-phase mode for air-cooled condenser line mass flow rate](image_url)

![Figure 11: Single to two-phase mode for air-cooled condenser line heat transfer rate](image_url)

Figure 10 shows the predicted and the calculated single to two-phase operating mode mass flow rate as a function of time. In the experimental model the pressure difference within the system between the heating and cooling section characterised the fluid driving force, and
hence defined the mass flow rate calculated in terms of the power input and the fluid properties at the inlet and outlet points of the heating section. It is seen that the predicted mass flow rate \( \dot{m}_{\text{predict}} = 38.32176 \text{ kg/s} \) increases rapidly to its maximum steady state value after 2.5 hours. The experimentally calculated mass flow rate \( \dot{m}_{\text{exp}} = 34.6754 \text{ kg/s} \) increases slowly to its maximum steady state after 4.5 hours. This is because the theoretical model set the power input to the system to its maximum value, while in the experimental model the power was gradually supplied to the heating elements. The result of this is that boiling starts earlier in the theoretical model than in the experiments.

Figure 11 shows the experimental and theoretical heat transfer rates as a function of time for the air-cooled condenser line. The accurately predicted electrical energy input to the system \( Q_{\text{elec}} = \dot{Q}_{\text{elec}} = \dot{V}_{\text{sup}} \cdot R_{\text{elec}} = 3000.33 \text{ W} \) is balanced by the heat output from the system through the air-condenser \( Q_{\text{acc}} = 2386.92 \text{ W} \) and the power lost in the system \( Q_{\text{line}} = Q_{\text{elec}} - Q_{\text{acc}} = 621.41 \text{ W} \). These results show that the heat added to the system by the heating elements through the fuel tank walls is carried out of the system by the air-cooled condenser, excluding some loses from the interconnecting pipe-work.

Figures 12 and 13 show typical sets of single to two-phase mode experimental and theoretical results for the heat exchanger line. Figure 13 shows an accurately predicted temperature rise of the working fluid in the heat exchanger. The \( T_{\text{heate}} \) increases rapidly to a maximum temperature of 92 °C after 2.585 hours, thereafter it decreases slightly to 90 °C after a further 2.625 hours. It is constant around 90 °C for about 50 min and then drops by 1 °C when steady state is reached around 89 °C in the top section of the heat exchanger. The \( T_{\text{heat}} \) increases in the same range as \( T_{\text{heate}} \), but reaches steady state around 90 °C, one hour earlier than \( T_{\text{heate}} \). The \( T_{\text{heate}} \) and \( T_{\text{heat}} \) oscillate steadily, with an oscillating amplitude of 1 °C and 1.25 °C, respectively. The \( T_{\text{heate}} \) and \( T_{\text{cool}} \) are accurately predicted and they oscillate around 61 °C and 33 °C with an oscillating amplitude of 3 °C and 2 °C, respectively, due to the mass flow rate oscillation.

Figure 13 shows that, the cooling process maintains \( T_{\text{cool}} \) around 193.54 °C. The \( T_{\text{cool}} \) and \( T_{\text{coolj}} \) were accurately predicted. They reach steady state earlier than \( T_{\text{heate}} \) and \( T_{\text{cool}} \). These results show that cooling the fuel tank by the heat exchanger keeps its temperature below 215 °C, when steady state is reached, as in the air-cooled condenser. This is due to a good average heat transfer coefficient of the heat exchanger compared to the air-cooled condenser. When steady state is reached a maximum amount of heat is added to the system. This offers much better cooling of the fuel tank.
Figure 14 shows the predicted and the calculated mass flow rates as a function of time. The predicted mass flow rate $m_{\text{left}} = 39.1645 \, \text{g/s}$, increases rapidly to its maximum steady state value after 2.8 hours. The experimentally calculated mass flow rate increases less rapidly to its maximum steady state of $m_{\text{left,exp}} = 40.054 \, \text{g/s}$ after 4 hours. Before steady state is reached, the theoretical and experimental calculated mass flow rates converge to the same values and increase with the same slope after 1.675 hours for about 1.078 hours. This is due to a well-predicted heat transfer coefficient that allows heat removal in the range of 2.0 to 2.117 kW in both the theoretical and experimental cases as seen in figure 15. The $m_{\text{left,exp}}$ oscillates early, with an oscillation amplitude of $0.1 \, \text{g/s}$ after 50 min. Thereafter, the oscillation amplitude increases to $2.2 \, \text{g/s}$ after 2.622 hours. This sudden increase of the oscillation amplitude shows a transition from single to two-phase flow. The $m_{\text{left,exp}}$ oscillation with an amplitude of $1.045 \, \text{g/s}$ starts 1.848 hours after $m_{\text{left,exp}}$. There may be three reasons for the late $m_{\text{left,exp}}$ oscillation. The first is the gradual power input to the system. The second is the liquid carried over in the vapour-liquid mixture flow. The third is the change in water level in the heat exchanger caused by the vapour-liquid mixture that condenses. This will cause the heat exchanger inlet thermocouple to be periodically immersed in hot water from the cooling jacket and then cooler water rising from the heat exchanger.

Figure 15 shows the heat exchanger line heat transfer rates as a function of time, calculated from theoretical and experimental results. It is seen that an accurate prediction of how the energy input to the system $Q_{\text{heat}} = 3008.53 \, \text{W}$ is balanced by the action of the heat output from the system through the heat exchanger $Q_{\text{he}} = 2111.92 \, \text{W}$ and the heat lost in the system $Q_{\text{loss}} = Q_{\text{heat}} - Q_{\text{he}} = 626.37 \, \text{W}$. There is still a slight deference between the power input to the system and the heat removed from the system plus the calculated heat lost ($Q_{\text{he}} + Q_{\text{loss}} = 2737.33 \, \text{W}$). This is for the same reason of additional heat lost from the system as explained in the previous paragraph. It is also seen that, the power output from the system by the water-cooled condenser is negative for the first 30 min. This is because when the process starts, depending on the cooling water mass flow rate, more heat is removed from the system than it is added. Although the same amount of heat is added by the heating element to the system through the cooling jacket, for the first 30 min the power output of the system is negative. These results show that the heat added to the system is recovered by the heat exchanger through the cooling water at a temperature of 36 °C.

Figures 16 and 17 show the heat pipe operating mode experimental and theoretical results for a 75% working fluid fill ratio using the air-cooled condenser line. When the loop is filled at 75% filling ratio the level of the working fluid is slightly above the cooling jacket. The fuel tank is entirely emerged in the working fluid. As seen in figure 15, results indicate that the $T_{\text{inlet,exp}}$ and $T_{\text{inlet,th}}$ predict well the experimental corresponding values $T_{\text{inlet,exp}}$ and
At steady state, the air cooled condenser outlet working fluid and coolant converge to the same values of 83 °C and 86 °C for the theoretical and experimental results. This is because the average heat transfer coefficient of the air-cooled condenser is much better in two-phase than in single and single to two-phase flow. Looking at figure 16 it is noticed that a good overall heat transfer coefficient for the air-cooled condenser in heat pipe operating mode accurately predicts $T_{\text{air}}$ and $T_{\text{cool}}$. This result keeps $T_{\text{wall}}$ around 193.84 °C. This is an improvement in the fuel tank cooling temperature compared to the single and single to two-phase operating mode.

Figures 18 and 19 show that, heat transfer coefficient is much better in the heat pipe operating mode than in single operating mode. The calculated mass flow rates increase and reach steady state fast as seen in figure 18. The heat transfer process takes place rapidly in two-phase due to a high heat transfer rate, dictated by a good heat transfer coefficient. This is also the reason why the mass flow rates are high and the calculated power well balanced in the system as shown in figures 18 and 19, respectively. In the time range 1 to 2 hours the predicted mass flow rate increases rapidly compare to experimental mass flow rate. This is again due to the gradual increase in the power for the experiments, as also outlined in previous paragraphs.

![Figure 16: Heat pipe mode with loop filled with 75% of working fluid air-cooled condenser line: 16) temperatures, 17) fuel tank wall and cooling jacket temperatures, 18) mass flow rate, and 19) heat transfer rate](image-url)
Figures 20 to 23 show experimental and predicted results of a typical set of heat pipe operating mode heat exchanger line temperatures for a 75% working fluid filling ratio. Figure 20 shows the heat exchanger inlet and outlet working fluid temperatures as a function of time. Figure 21 shows the heat exchanger inlet and outlet cooling water temperatures as a function of time. Figure 22 shows the cooling jacket inlet and outlet working fluid and fuel tank temperature as a function of time. The heat exchanger is supplied with a steady stream of water and the electrical power is switched on. Boiling is seen to start at about 45 min after the power is switched on, as seen by the characteristic start-up mass flow rate transient of up to 400 g/s, whereafter it quickly settles down to a reasonably steady 100 g/s, after the shut-off valve is closed (see figure 21). As seen in figure 21, experiments indicate that when boiling starts, the inlet heat exchanger temperature increases rapidly to a maximum value of 100 °C then decreases to 78 °C before steady state is reached at 90 °C. This is because when the experiment starts, the shut-off valve for air released is opened to release all the air from the system by boiling. Boiling starts at 100 °C, which is the saturated temperature corresponding to the atmospheric pressure. Figures 21, 22, 23, 24 and 25 all show a reasonable correspondence between the theoretical and the experimental results. This is especially so when a steady heat pipe operating mode has been achieved.

5 Conclusions and recommendations

The objective of this paper was to experimentally evaluate and demonstrate the validity of the theoretically model simulating a waste heat removal and utilisation for a G-IV reactor spent fuel storage tank system. Cooling was selected as a mechanism of waste heat recovery and utilisation process of a G-IV reactor spent fuel storage tank. Heat was then passively removed from the fuel tank by the heat exchanger and/or air-cooled condenser, and then carried out of the system through the heat exchanger cooling water. Hence, the designed G-IV reactor spent fuel storage tank WHR&US was found to function well. This design used two incoloy electrical heating elements of 3 kW each to simulate the fuel storage tank waste heat. The as-tested scale model of the used and/or spent fuel tank (shown in figures 1 and 2) has four shut-off valves to manually regulate the two lines of the WHR&US, instead of an automatic thermostatic control valve. The later will automatically divert the steam from the cooling jacket to the secondary line should a fault or a stoppage occur in the primary cooling
line as a result of a mechanical failure or planned maintenance of the power conversion system. An automatic thermostatic control valve requires specific analysis which depends upon the thermosyphon thermo-dynamic flow response. Use of a domestic electric hot-water geyser thermostatic control valve was considered. It was however unsuitable because its over-temperature option opens at about 93 °C, whereas the design specification for the cooling jacket is 250 °C. This requires the WHR&US loop to be established first, and thereafter the thermostatic control valve design options may be investigated in accordance of the loop thermo-dynamic response. It is thus recommended that such a thermostatic control valve be researched in future studies.

References
Dobson RT. 2007. Advanced heat transfer course, two-phase flow and heat transfer. Stellenbosch University, Stellenbosch.

Acknowledgement
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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>Cross-sectional area, m²</td>
</tr>
<tr>
<td>b</td>
<td>Fin length, m</td>
</tr>
<tr>
<td>h</td>
<td>Boiling number</td>
</tr>
<tr>
<td>c</td>
<td>Compression corrective factor</td>
</tr>
</tbody>
</table>

Heat transfer area, m²
Specific heat, $k_f/k_g \cdot R$
Coefficient of friction,
Convective number
Diameter, $m$
Enhancement factor
Froude number
Friction factor coefficient
Form factor
Acceleration of gravity, $m_f k_e^2$
Grashof number,
Heat transfer coefficient, $W/m^2 \cdot K$
Specific enthalpy, $k_f/k_g \cdot K$
Latent heat of vaporisation, $k_f/k_g$
Minor loss coefficient
Thermal conductivity, $W/m \cdot K$
Working fluid mass, $kg$
Mass flow rate, $kg/s$
Number
Number of fins
Nusselt number, $Nu = h_d/k$
Pressure, $N/m^2$
Perimeter, $m$
Prandtl number, $Pr = \mu e_p/k$
Thermal power, $W$
Rayleigh number, $Ra = GrPr$
Reynolds number, $Re = \rho d_L v / \mu$
Superscripts
$t$ Time step $t$
$\Delta$ Change of time
Subscripts
$a$ Air, ambient
$ace$ Air-cooled condenser
$ac$ Air collector
$atm$ Atmospheric
$bu$ Bubble suppression
$c$ Characteristic
$cb$ Convective boiling
$cl$ Cooling jacket
$co$ Condenser
$conv$ Convection
$cu$ Copper
$cw$ Cooling water
$d$ Diameter
$df$ Drag friction
$elek$ Electrical
$eq$ Equivalent
$ev$ Evaporator
$f$ Fluid, factor, fin
$frict$ Friction
$ft$ Fuel tank
$g$ Gas
$h$ Homogenous, hydraulic
$he$ Heat exchanger
$hs$ Heat source
Greek letters
$\alpha$ Void fraction
$\beta$ Thermal expansion coefficient, $R^{-1}$, fin parameter
$\epsilon$ Emissivity, Volume void fraction
$\theta$ Angle
$\nu$ Dynamic viscosity, $Ns/m^2$
$\nu_p$ Kinematic viscosity, $m^2/s$
$\nu_p$ Poisson’s ratio
$\rho$ Density, $kg/m^3$
$\rho_p$ Homogeneous density, $kg/m^3$
$\sigma$ Stefan- Boltzmann constant, $W/m^2 \cdot K^4$
$\tau$ Shear stress, $N/m^2$
$\tau_2$ Two-phase multiplier
$\psi$ Prandtl number function
Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>DFAC</td>
<td>Dual function absorption cycle</td>
</tr>
<tr>
<td>FHSS</td>
<td>Fuel handling and storage system</td>
</tr>
<tr>
<td>GTHR</td>
<td>Gas turbine helium reactor</td>
</tr>
<tr>
<td>HTR</td>
<td>High temperature reactor</td>
</tr>
<tr>
<td>ORC</td>
<td>Organic Rankine cycle</td>
</tr>
<tr>
<td>RCCS</td>
<td>Reactor cavity cooling system</td>
</tr>
<tr>
<td>US</td>
<td>United States</td>
</tr>
<tr>
<td>WHRS</td>
<td>Waste heat recovery system</td>
</tr>
<tr>
<td>WHR&amp;U</td>
<td>Waste heat recovery and utilisation system</td>
</tr>
</tbody>
</table>

Bibliography

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Robert T Dobson: is a Stellenbosch university senior lecturer and chairman of the Stellenbosch university nuclear engineering safety system. He was Franck M Senda study leader when they took a project on aspect of waste heat recovery and utilisation in nuclear technology.